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THE NATION'S LABORATORY FOR ADVANCED AUTOMOTIVE TECHNOLOGY

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Test and Evaluation of the

LMTV Driveline

1998

By Stephen J. Aamodt

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<p>A Light Medium Tactical Vehicle (LMTV) was tested on a chassis dynamometer at the US Army Tank Automotive Research, Development and Engineering Center (TARDEC) test labs from the months of August through October of 1998. The purpose was to determine the strains and accelerations in the driveline system and to determine the effect of different driveline configurations on these values. Tested variations included the OEM 3.5 inch diameter rear driveshaft, a larger 4 inch diameter (A1) rear driveshaft, a straightened and balanced OEM driveshaft and a Michigan Scientific Driveshaft modified to eliminate hinging. Other driveline changes included replacing the OEM flywheel with a nodular iron version and replacing the yokes utilizing straps, to hold in the driveshafts to full rounds yokes, and lastly using nylon versus metal thrust washers in the universal joint end caps. Results showed that the replacement components, namely the 4 inch diameter driveshaft with nylon thrust washer and full round yokes, and the nodular iron flywheel housing, yielded more favorable results than the original equipment.</p>			
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INTRODUCTION

The Light Medium Tactical Vehicle (LMTV), a member of the Family of Medium Tactical Vehicles (FMTV) was experiencing failures with the cracking of the flywheel housing and a breaking apart of the rear transfer case u-joint/driveshaft coupling. In July of 1998, the FMTV PM office approached TARDEC and requested help in determining a solution to these problems.

The initial purpose of testing was to determine the reason for the driveshaft and flywheel housing failures. Follow on tests included runs to validate a Michigan Scientific Corporation drivetrain model and a 6000 mile durability and validation test on the proposed vehicle fix.

The vehicle was to be tested utilizing a two roll chassis dynamometer that allowed both front and rear tires to be driven. The vehicle was set-up and instrumented in TARDEC's building 212, test cell 9. Personnel from the Mobility Test Operations Team, the Physical Simulation Team, and the Propulsion Products Support Team conducted the testing. The following report includes details of test objectives, set-up, discussion of results, and conclusions and recommendations for this vehicle.

OBJECTIVES

There were two initial objectives of this test program. The first was to determine the stress levels occurring in the flywheel housing due to various configurations of the rear driveshaft and at various propshaft angles. The second objective was to determine accelerations at various points throughout the vehicle's drivetrain and compare them to those found by utilizing different driveshafts and various driveshaft angles. A test plan was laid out in order to achieve these goals. Refer to Appendix A for the test plan. Additional objectives were added as testing continued. One was the utilization of the data obtained to validate a drivetrain model that was developed by Michigan Scientific Corporation. Also, a 6000 mile validation test on the proposed fix (nodular iron flywheel housing, A1 shaft, full rounds yokes) to the vehicle was requested.

TEST EQUIPMENT

LMTV1:

Manufacturer: Stewart & Stevenson
Vehicle S/N: AT7378BCMF
Production Date: 12/97
Initial Mileage: 57 miles

LMTV2:

Manufacturer: Stewart & Stevenson
Vehicle S/N: AT0072B-EB
Production Date: 6/93

Initial Mileage: 10,418 miles

Dynamometer:

Manufacturer: Clayton Industries
Model: CD1400
Absorption Units: 50 HP each axle

RESULTS AND DISCUSSION

LMTV 1 arrived at TARDEC on July 22, 1998. The vehicle had 57 miles on it. The vehicle was moved to Building 212, test cell 9 for testing.

A vehicle that runs on a chassis dynamometer needs to have shaved tires, if running for any duration is desired. This is because the friction caused by the tread moving around and squishing together builds up heat faster than using tires with no tread. An overheated tire could result in a blowout during testing. The tires were, therefore, removed and sent out for shaving.

The vehicle was instrumented per the list in Appendix A. The shaved tires were remounted and the vehicle was installed on the chassis dynamometer where the final instrumentation could then be performed.

Prior to testing, the vehicle cooling fan clutch was locked in the 'on' position. This vehicle contained an older version of the cooling fan and its clutch. By locking it on, this kept the fan running full time and reduced the vibrations associated with the clutch engaging and disengaging as would otherwise have been experienced.

To determine a baseline, the rear driveshaft was removed and sent out to Joint, Clutch & Gear Service, Inc to check the straightness and balance. It was found that, compared to their standards, the shaft we had was both out of round and unbalanced. Refer to appendix B for the message relaying these facts. Joint, Clutch & Gear Service, Inc mistakenly balanced and straightened the shaft prior to sending it back to us.

When checking with the PM's office regarding this finding, it was revealed that the shaft did meet the specifications for balance as provided by the manufacturer, Meritor, but did not meet the specification for runout. The Meritor specifications stated for balance that a value of 2.06 inch-ounces is the maximum on the slip end and 1.49 inch-ounces is the maximum on the weld yoke end. Runout values that should not be exceeded are 0.005 inches on the spline plug, 0.015 inches in the center and 0.010 inches on the weld side. Refer to figure 1 for the visual locations of these values as specified by the manufacturer.

Due to the findings on the rear driveshaft, we decided to remove and check the front driveshaft. We found that the U-joint on the transfer case side (slip joint end) was tight

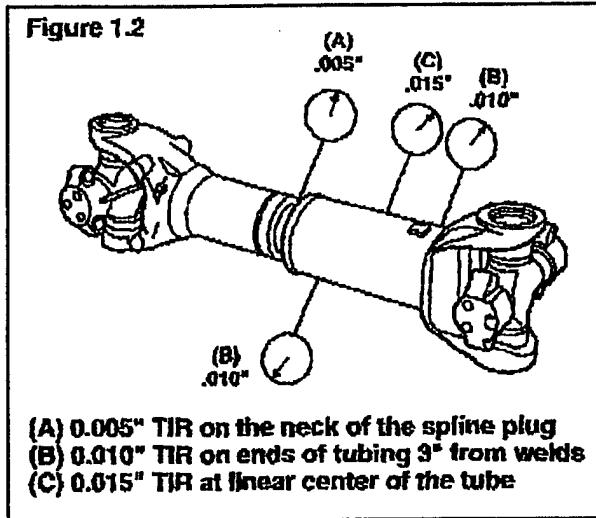


Figure 1 - Driveshaft Runout Specifications

and did not pivot easily. Releasing the torque on one of the end caps allowed smooth rotation to occur. This seemed to indicate a clearance problem. Due to this finding we decided not to have this shaft checked for balance or roundness. We later found out that, according to Meritor, a tight u-joint is not necessarily bad, so long as the motion is smooth and it does not jerk from point to point. The PM agreed that we should test with a typical production shaft (versus a straightened and balanced shaft), and as such would ship 3 more sets out to us. We would pick the ones measuring closest to 1.0 inch ounce balance on each end and use these as this is the typical production value.

The first set that arrived also had a tight u-joint on the short prop shaft, slip joint end. Both driveshafts were sent out to Joint, Clutch & Gear Service, Inc to check the balance and runout. Both shafts were within Meritor's balance specification, but neither meet the specification for runout. The results of this inspection can be found in Appendix B. Upon consultation with the PM's office it was decided to run using the Joint Clutch balanced rear driveshaft and the, just arrived, front driveshaft. The original front driveshaft was sent to Meritor for inspection on the tightness in the u-joint.

Checkout testing began on August 1, 1998 and actual testing started on August 4. The vehicle configuration included a production flywheel housing and a balanced and straightened driveshaft with metal thrust washers in the end caps. The vehicle, as installed on the chassis dynamometer, is shown in figure 2. The first test conducted was a motoring test to determine the rolling resistance the vehicle incurred by being on the chassis dynamometer. A motoring test means that the dynamometers are speeding up the rolls and hence the vehicle while the vehicle's engine is off and the transmission is placed in neutral. The value found turned out to be just over 30 lbs/ton. For a wheeled vehicle, this is indicative of travel on a decent paved road.

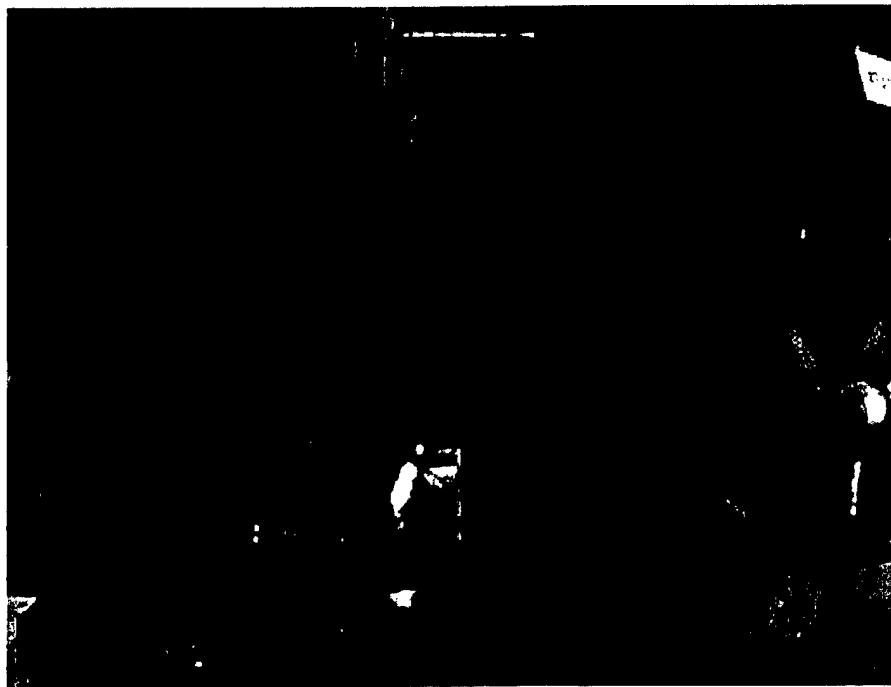


Figure 2 – LMTV 1 on Chassis Dynamometer

Following the motoring tests the first vehicle test was run in 2 wheel drive mode with speeds up to 60 mph. The rolling resistance found from motoring had not determined prior to this test so effectively this test was run at a rolling resistance of just about 60 lbs/ton. This rolling resistance is more indicative of travel on a dirt trail. The driveshaft angle for this test was 8.5°. This reflects the vehicle in a normal state with no payload in the bed. Testing was also conducted with a payload of 5000 pounds in the bed reflecting the stated maximum carrying load of the vehicle. The driveshaft angle found for this configuration was 7.3°.

The first failure on this vehicle occurred the second day of actual testing. The vehicle's water pump cracked and started leaking coolant. Testing was stopped with 80.3 miles on the vehicle. The water pump was found to have a 5 ½ inch crack starting across the top and moving down the side (see figure 3). The location of the crack indicates that vibration in the vertical direction is the most likely cause.

It is our opinion that some redesign of the bracketry holding up the alternator would likely reduce cracking of the water pump. The water pump is bolted to the engine on the left side (referring to figure 3). Bolted onto the right side of the water pump is the bracketry used to hold up the alternator. The weight of the alternator and its associated bracketry is just over 60 pounds. This weight is supported by a combination of the water pump and a straight bracket off the engine which is in turn attached to the U-bracket. However, due to the configuration of this bracketry (see figure 4), the water pump ends up supporting a good portion of this weight.



Figure 3 – Cracked Water Pump

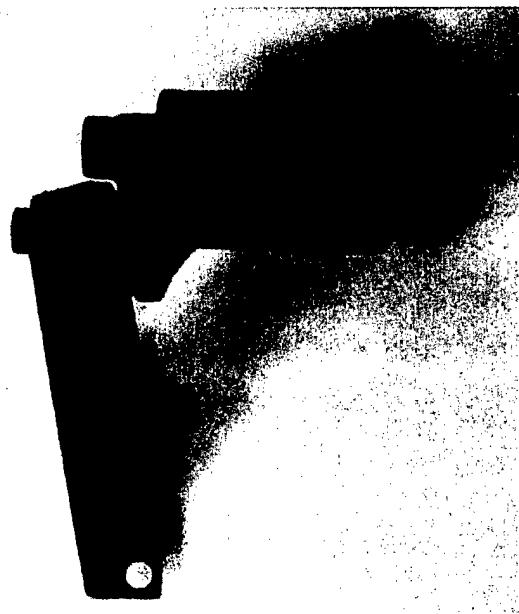


Figure 4 – Alternator Support Bracketery

Notice that the straight bracket connection to the supporting the U-bracket is such that it effectively forms a cantilever beam. This essentially leaves the far end, where much of the alternator weight resides, to flex and bend freely much as a weight placed on the end of a meter stick would react. The U-bracket bolts to the water pump in order to stop this from occurring. However, as the vehicle drives, the forces being exerted are still there which continually stress the water pump. Over time, these forces cause the pump casing to wear and potentially, as our case illustrates, fail. Of course this is dependent on many factors such as how severe the forces are that are imposed, such as hitting large bumps, and whether particularly destructive resonant frequencies are encountered. While it is possible that this failure would have occurred regardless of vehicle configuration, just looking at the supporting bracketry indicates that a more sturdy solution exists that may reduce the chance of random failures.

In checking over the vehicle, a loose deaeration hose located on top of the engine was found. It was not off, but the screw had loosened enough to allow it to slide around freely.

While preparations were being made to re-install some strain gages it was found that the starter motor was cracked in the casting around the top four bolts that connect it to the flywheel housing. The portion that cracked off is shown in figure 5. 4 of the 5 bolts used to hold the starter motor to the flywheel housing had cracks completely around them.



Figure 5 – Starter Motor Crack

Inspection of the starter motor found that rust was present in the cracks. Finding rust indicated that this condition had been present for a while. Since this vehicle had just over 80 miles on it at this time, it is possible that this particular casting may have had imperfections in it that could have caused it to fail soon after leaving the assembly plant.

Once these items were replaced, testing began again only to notice immediately that the rear driveshaft was wobbling. One end cap on the transfer case u-joint had overheated and seized up. Inspection revealed that this failure was typical of those witnessed in the field. It is suspected that the reason this was not noticed before was do in part to thermal expansion that, when the end cap failed, caused the u-joint to be held relatively in place until it cooled off. While the shaft may not have been displaying wobbling that could be visually seen, it was likely causing excessive vibration throughout the rest of the system. It is probable that this extra vibration helped cause the premature failure of the water pump.

The wobbling itself was being caused by the space created between the end cap and the yoke. From figure 6, the amount of play that was discovered was such that a quarter would easily fit in this space. Also visible is the scoring in the metal, caused when the cap overheated, and a slight bulge in the center of the cap. Though some needles were found that were a little squished, nothing was wedged underneath the cap. The space between the yoke and end cap allowed the driveshaft to shift off center which, when spinning, displayed the wobbling motion. This failure occurred while using the Joint Clutch & Gear straightened and balanced shaft. Later analysis of the shaft indicated that it had bent drastically when the failure occurred. Refer to Appendix B for the results.

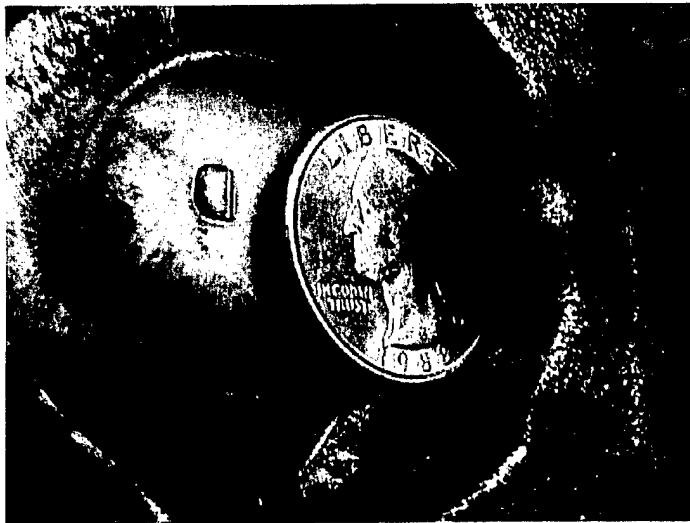


Figure 6 – Play Between the End Cap and Yoke

At this point Joint Clutch & Gear sent a representative out to evaluate the cause of the failure on the driveshaft they had balanced and straightened. Upon inspection of the system, the representative stated that propshaft life cycle prediction with this arrangement was very difficult and failures could occur at any time. This was based on the angle of the shaft and the fact that the transfer case and axle are parallel to each other but have an opposite slope to that of the driveshaft. He also commented on whether a larger shaft was likely to solve the problem. He said that having a bigger shaft, to include a bigger and longer spline, may last somewhat longer but would eventually fail. He recommended tilting the transfer case to a flatter, if not opposite, slope. The axles should remain

parallel in slope to the transfer case. This recommendation was made even though he was aware that adverse affect on the front shaft would result.

A replacement driveshaft was received that had nylon thrust washers in the transfer case side instead of the metal thrust washers that were, at that time, used in production. The nylon is purported to have better properties with resistance to wear and failure. The straps were also replaced on the chance that they might have been damaged.

Testing resumed and runs in two wheel drive mode with speeds up to 60 mph were recorded. The four different angles that the testing was performed at included a simulated jounce condition, the vehicle with a 5000-pound payload in the bed, the vehicle with no payload, and a simulated rebound condition. The effective angle at the transfer case universal joint consists of the driveshaft angle plus the angle of the transfer case since these are not parallel but are actually on opposite slopes. Figure 7 breaks down these angle comparisons.

Condition	Driveshaft Angle	Transfer Case Angle	Effective Angle
Rebound	9.7	2.4	12.1
No Payload	8.7	2.4	11.1
5000 lb Payload	7.2	2.4	9.6
Jounce	6.6	2.4	9.0

Figure 7 – Driveline Angles

The data indicated that the vertical, lateral and longitudinal accelerations all increase as the driveline angle increased. The root mean square method showed acceleration that increased anywhere from 50% to 280% between the jounce and rebound conditions.

When the vehicle had 163 miles on it a marked reduction in strain for gage 1 on the flywheel housing was found. Upon visual inspection, a crack was found across strain gage 1 on this production housing. This crack is shown in figure 8.

This crack is in the same location as those routinely witnessed in the field. While this failure does not keep the vehicle from running, the flywheel housing should still be replaced. The crack has the potential to grow and cause a much more catastrophic failure if not addressed.

At this point the data promised to the PM's office was collected and delivered for their use. Up to this point, determination had been made that accelerations at the transfer case rear universal joint did increase as the propshaft angle was increased. There are several solutions to resolving this problem. Tipping the transfer case backward, or at least straight, and making the rear axle parallel to it would reduce the effective angle and, therefore, decrease the effective angle and hence accelerations at the transfer case rear



Figure 8 – Crack on Production Flywheel Housing, Strain Gage 1

universal joint. However, this solution would cause the effective angle of the front driveshaft to increase. Another solution would be to use the old style axle with a top input for the driveshaft. This would decrease the effective angle. This solution is currently being evaluated at Aberdeen. A final means of reducing the angle would be to lower the powerpack several inches. It is felt that the utilization of a constant velocity (CV) joint may not completely solve the problem either. However, a CV joint is likely to last longer.

A new flywheel housing was delivered to replace the cracked one. It was a production housing with no miles logged on it yet. The engine was removed from the vehicle, the flywheel housing replaced, and the engine re-installed.

Since we were witnessing a good level of heat buildup in the tires for extended runs we decided to reduce the tire pressure on the dynamometer rolls. This would create less tire friction, and therefore reduced tire heat buildup. We checked with the PM and Meritor to determine whether there would be any problem with slightly raising the front and rear axles to relieve a little more pressure from the tires. The conclusion was that no adverse drivetrain problems should result and this was acceptable. Therefore, as visible in figure 9, blocks were placed under each axle so the contact area between the tires and rollers could be reduced to a level that would allow extended vehicle running without overheating tire temperature. Fans, which had been used before, were still used to aid cooling.

Testing resumed with the new flywheel housing. Tests at various steady state speeds and at various conditions were performed. Acceleration sweeps were also run which could be used to indicate, at what speed in particular, peaks may be occurring within the system. The purpose of many of these tests was to help in the validation of the driveline model that Michigan Scientific Corporation was working on.

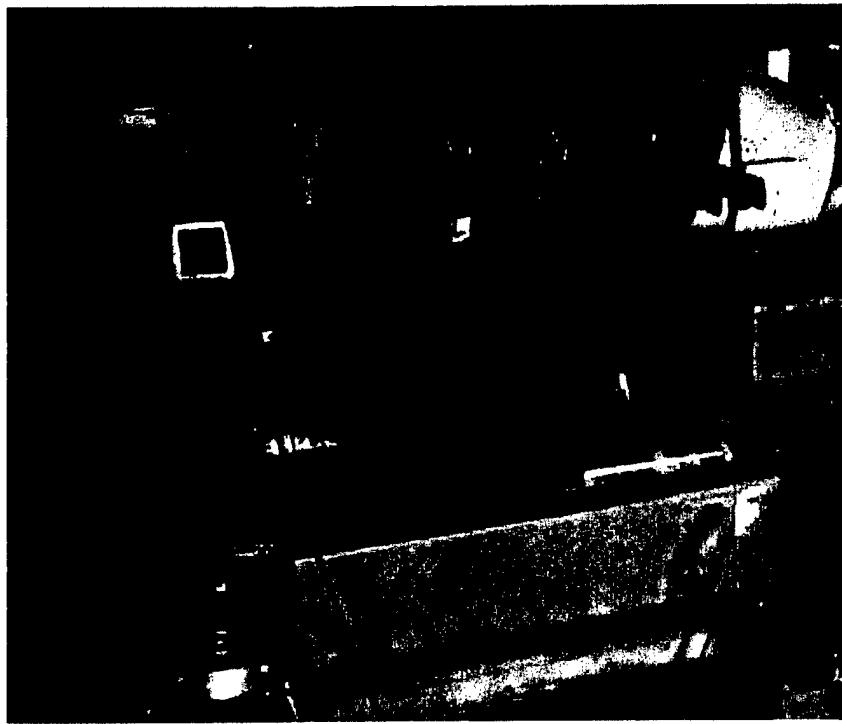


Figure 9 – Tire and axle orientation on chassis rolls with I-beam support

On August 28, Michigan Scientific delivered a modified driveshaft that was stiffened to reduce, and hopefully eliminate, the hinging effects (see figure 10). Hinging is what occurs when the two ends of the driveshaft bend in opposite directions. It is determined by the distance between the spline contacts. As seen in figure 10, the shaft has a tube over both sections, welded to the long with a tight fit on the short end. This keeps the shaft from hinging in the spline.

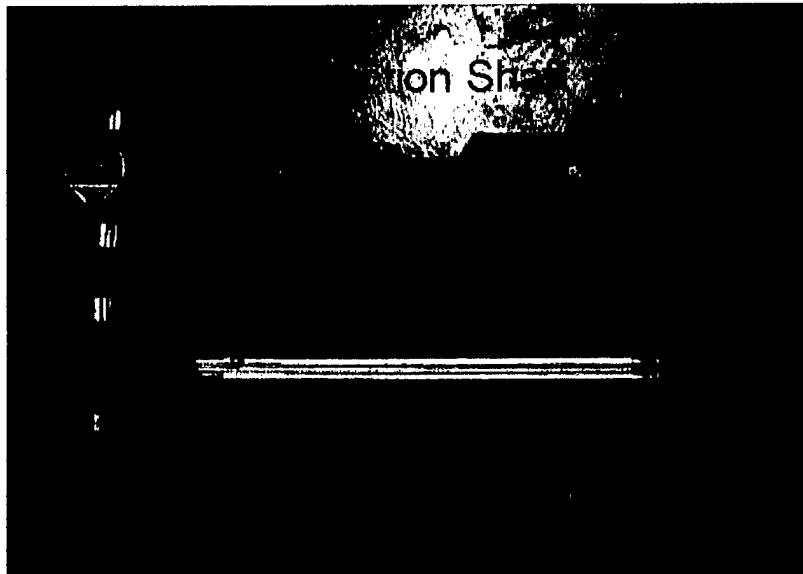


Figure 10 - MSC shaft eliminating hinging

The production driveshaft with nylon thrust washers was removed and examined. It had approximately 150 miles on it at this time. The universal joints spun freely and visual inspection revealed nothing abnormal.

Comparison runs were performed with the MSC driveshaft, nylon thrust washer production driveshaft, and also with no driveshafts installed at all. Similar test points were run in each configuration for comparison validity. The data collected could be used to compare the performance difference between the three configurations.

The data for both the MSC driveshaft and the production driveshaft indicated marked increase in acceleration levels as the driveline angle increased. The MSC driveshaft, with the elimination of the hinging effect, proved more effective than the production driveshaft with nylon thrust washers in the resonant frequency range at the high angles. Outside of the resonant range and high angles, no particular driveshaft had an advantage all the time. Figure 11 shows a visual representation of the performance of these two shafts within a resonant frequency range measuring g-loads from peak to peak.

The flywheel housing strain, measured at strain gauge one, does not appear to be effected by drivetrain angle. What does affect the flywheel housing strain appears to be the speed at which the vehicle is moving. From figure 12, it is visible that this type of relationship exists.

Reduced speed then is the key issue behind keeping the strain levels down on the flywheel housing while a reduced driveline angle is the key to keeping the accelerations down at the transfer case universal joint.

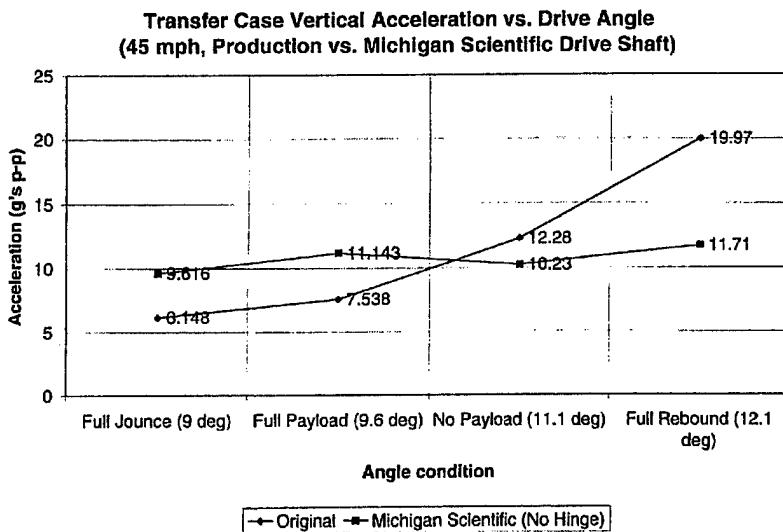


Figure 11 - Comparison of Vertical Accelerations in Resonant Range

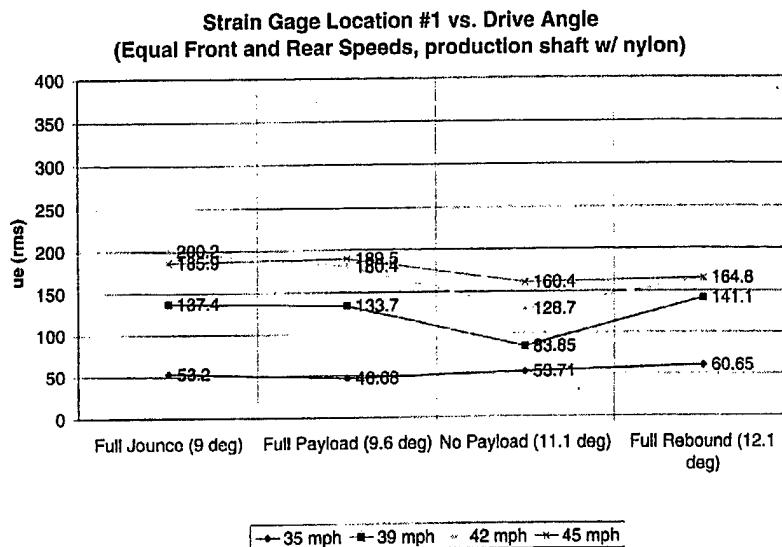


Figure 12 - Production Flywheel Housing strain versus speed and angle

On 3 September, the Red Team (previously formed to resolve the LMTV driveline issues) decided that TARDEC would run a 6000 mile validation test using the nodular iron flywheel housing and the full round, 4 inch, driveshaft. Initial testing already performed at Aberdeen using this configuration was having good results. These components were to be shipped to us in the near future.

The Caterpillar nodular iron flywheel housing arrived on 4 September. The vehicle was removed from the test stand to have the engine pulled in order to replace the flywheel housing.

Testing to this point had relied on the operator's ability to equalize front and rear speeds manually. This is not automatic in 2x4 mode because the torque split is 70% on the rear and 30% on the front. Allison Transmission was contacted regarding a method to lock the transfer case into full-time 4x4 mode (equal speeds on front and rear axles). They provided a solution that involved splicing into the electrical connector going to the transfer case. The power wires in the connected were bypassed and wired to the vehicle's batteries. An intermediate switch was used to allow the 4x4 mode solenoid to be energized when the switch was engaged by the test operator. This would allow the 4x4 mode solenoid to be energized the whole time the vehicle was running. This would allow the vehicle to be tricked into 4 wheel drive at all speeds, not just those below 40 mph. This was incorporated into the vehicle while the engine was being reinstalled.

The nylon thrust washer driveshaft was again installed on the vehicle. Instrumentation checkout was run once the vehicle was re-installed on the chassis dynamometers. Following the instrumentation checkout, a steady state run with 5 mph increments from 20 mph up to the governed speed (58 mph) was run using the 4x4 mode lockup circuit. The 4x4 mode circuit was successful for about two hours, however, while idling before running the acceleration sweep the wiring for the 4x4 mode circuit melted through by the switch.

Checking the wiring showed heat damage along most of the external portion we had added in. When checked, the transfer case external electrical connector indicated only a minimal resistance that hinted that the solenoid had shorted out. Removal of the solenoid showed us the cause of the short. The connector on the outside of the 4x4 mode solenoid had apparently shorted and melted, causing the rest of the circuit to go. The solenoid itself was still functional but the connector produces intermittent failures when jiggled. Since this circuit was a modified one, this failure would not have occurred in the field. We were still able to run 2x4 mode and equalize the tire speed until a new solenoid was received.

Later consultation with Allison revealed that this was the first time this had been tried. With the results that occurred they suggested that drawing 12 volts from the batteries and then grounding to the vehicle may have caused an increased current draw through the circuit that could have caused the short. The proposed solution was to completely disengage the 4x4 engagement circuit from the vehicle by disconnecting the transfer case wiring harness and providing a completely separate 12 volt and ground source. The speed sensors could be jumped or could be kept connected through extension wires once the connector was pulled. Later trials showed that jumping the speed sensor caused the transmission to just lock up and not shift. Using extension wires proved successful. This method ran successfully until testing was stopped.

When the new A1 shafts and full round yokes arrived they were immediately installed on the vehicle. An A1 shaft is 4 inches in diameter, has a longer spline engagement, and is more precisely balanced. The accompanying yokes are full round, meaning they no longer require straps to bolt the driveshaft in place. Since time was a constraint these shafts were not sent out to have the balance and straightness checked.

During the first phase of the 6000 mile durability test the vehicle was run with the transmission oil roughly two gallons short. Some oil had been drained from the transfer case so that the yokes could be installed, but was not replaced afterwards. The oil was checked before, but not after the vehicle was started up. The oil only circulates between the transfer case and the transmission while the vehicle is running. The vehicle odometer, prior to start of testing, read 415 miles.

The night crew experienced four overheats of the transmission oil indicated by the light in the cab. The first occurred at 0237 hours when the oil temperature hit 354°F. The vehicle was cooled down for 37 minutes to a transmission oil temperature of 237°F and restarted. At 0321 hours they stopped again with a transmission oil temperature of 335°F. Before restarting, the vehicle was cooled down for 49 minutes and the transmission oil temperature dropped to 245°F. At 0417 they shut down again with a temperature of 338°F and let the vehicle cool for 51 minutes when the oil temperature dropped to 175°F. The test was aborted at 0527 hours with a temperature of 349°F.

Upon arrival of the day shift, the oil shortage was noted. There was still oil in transmission sump but the level was around 2 gallons low. The vehicle odometer read 677 miles. The vehicle was checked for damage. The transmission and transfer case oil was drained and checked for metal shavings. Some tiny flakes were found but these are

fairly typical, especially during the first transmission oil change. The oil was replaced with new oil and a check out run was performed to verify all gears were working. According to the vehicle operator, the vehicle operated as it had in the past and the transmission shifted up and down through all the gears normally. A check with Allison indicated that the transmission may or may not be damaged but the fact that it shifted normally was a good sign.

A no payload run was then performed, as a checkout to make sure everything was functional and since this data would be useful for comparison purposes. Testing was aborted when we noticed several accelerometers had failed. The heat generated from sustained testing had exceeded the accelerometers maximum temperature limit and caused them to fail.

While working this issue a new 4x4 mode solenoid arrived. When the transfer case valve cover was removed to replace the old solenoid with the new one, many large metal shavings and pieces were found. These were primarily blocking the third port down on the rear of the solenoid. Draining the transfer case oil revealed more large shavings. As these were not present when the oil was first drained after the overheating occurred it seems the failure occurred during the no payload checkout run. The transmission oil was also drained, but revealed only a few tiny shavings, which is relatively normal.

It is very likely that the low oil level, and the excessive heat buildup caused by it, lead to a delayed failure in the transfer case. Later disassembly of the transfer case/transmission components halves showed that the screen used to filter oil between the transfer case and transmission caught many particles. The hinging one the rear driveshaft was checked and it proved to be the same as before the overheating occurred.

Another LMTV (LMTV2) was picked up from the Walboro Corporation in Auburn Hills and delivered to TACOM. This truck had 10,418 miles on it. The idea was to take the transmission and transfer case from this truck and install in into ours. However, since LMTV2 was produced, a new transmission and transfer case layout was introduced, which our original truck (LMTV1) had. This made the two incompatible. Consequently, we ended up using the LMTV2 chassis and putting our original instrumented engine into it. The reason for this was that it was already removed from the truck and was less likely to have problems than the LMTV2 engine, which had over 10,000 miles on it.

The 6000 mile durability test was re-started. The hinging was recorded to be 0.010 inches. After 173 miles of running the hinging had increased to 0.015 inches.

After running 270 miles oil was found on the engine were it should not have been. It was finally traced back to a crack in the engine front cover just above the compressor. It is a hairline crack and difficult to see. The arrow in figure 13 shows the location of the crack.

Because of our recorded increase in hinging, Meritor came out to perform a hinging measurement on the rear driveshaft and see if it was still in good shape. The hinging value came in between 0.011 inches and 0.012 inches. Heat does not affect the hinging measurements so we must have been doing our measurements inaccurately.

The cause of the crack in the front cover was found. Per advice by Matt Townsend, we checked and found a bolt missing that connects the compressor and power steering pump to a mounting bracket attached to the engine. Refer to figure 14. Without this bolt in place, a cantilever beam is formed with the weight of the compressor and steering pump being held solely by the bolts on the engine cover. Vibration is allowed in both the vertical and lateral directions.

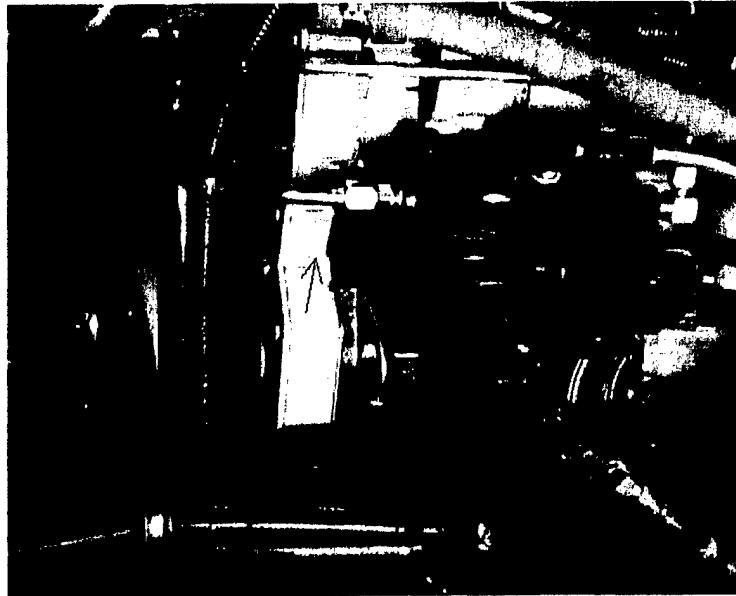


Figure 13 – Crack in the Engine Front Cover

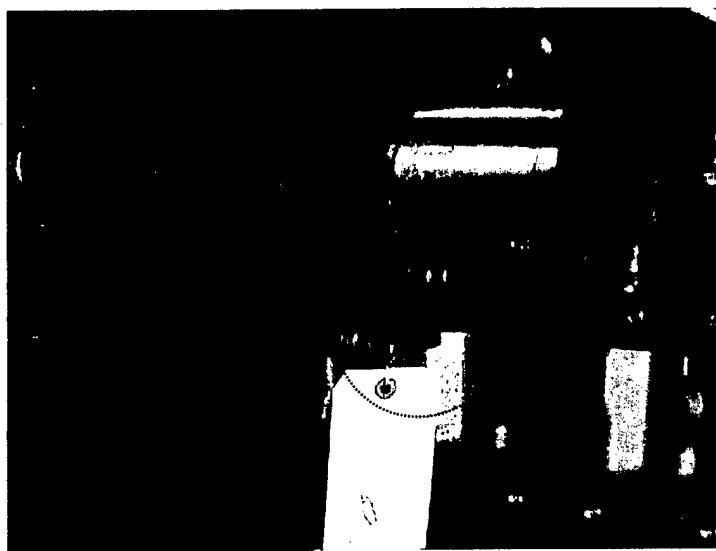


Figure 14 – Location of Missing Compressor/Power Steering Pump Bolt

It is not known whether this bolt fell out at some point or whether it was never installed. We suspect that it vibrated out during testing. Securely locking the bolt in place, by using a locking nut or self-locking bolt, should resolve problems like this in the future.

The engine with the cracked cover was replaced by the original engine from LMTV2. Continuation of the 6000 mile durability test then resumed. Testing continued until the PM's office requested the test be terminated. The vehicle had completed 1382 miles of durability testing. The driveshaft and the flywheel housing had not failed and were performing successfully at this point.

A final hinging measurement showed a maximum hinging of 0.0155 inches. Meritor did not check the hinging for us this time. Keeping with just the hinging number we measured then, over the 1382 miles the driveshaft hinging rapidly increased 0.005 inches but then only increased 0.0005 inches for the remainder of the test.

CONCLUSIONS AND RECOMMENDATIONS

As per the initial purpose of this test, it was indeed found that the driveline angle had an effect on the accelerations in the system. At times, the rebound angle tested could generate accelerations almost three times larger than the jounce angle tested. Incorporation of a top input rear axle would be extremely useful in this respect. Another way to accomplish the angle reduction would be to drop the powerpack several inches. Finally, tilting the transfer case to a flat angle (versus the 2.4 degree forward tilt currently used) would help the rear driveshaft problems. Note, however, that this solution would have an adverse impact on the front driveshaft.

Utilization of a larger diameter driveshaft would probably increase driveshaft and its accompanying universal joints life. However, this may just delay the problem without actually resolving it. Incorporation of a constant velocity joint could prolong life. Testing of this option would be required to prove this. Additionally a constant velocity joint may have additional heat related issues because they typically generate more heat during operation.

Laboratory testing acts as an accelerated means of determining what will happen in the field. This is so because of the ability of a lab to test at steady state points which can emphasize trouble points that a fielded vehicle may go through but only transiently. As such, failures of lab tested components generally indicate that similar field failures can be expected, though likely later in the components life.

For this vehicle, there is concern over several additional aspects that were not specifically addressed during testing. In addition to the driveshaft and flywheel housing field failures that we were testing for, our testing revealed several other failures that should be addressed.

As mentioned in the discussion, the bracketry holding the alternator is questionable. Its ability to support its own weight and that of the alternator without adversely impacting

the water pump housing does not appear sufficient. The moments added to the bracketry by the moving vehicle could be effectively causing an early failure in the water pump housing. A redesign of this bracketry will help alleviate this problem.

The bolt attaching the compressor and water pump to its engine mounting bracket likely vibrated out during testing causing the engine front cover crack. Refer to the discussion section for the specifics. In order to keep this from happening in the future we recommend using a self-locking bolt or a locking nut in this location.

APPENDIX A
Initial Test Plan and Instrumentation List

LMTV Test Plan
Revised 8-7-98

Objective:

To take vehicle driveline vibration and temperature measurements against a variety of suspension and drivetrain geometry's in order to verify causes of failure and validate proposed fix.

Instrumentation Points:

Vibration Instrumentation:

Channel	MegaDac #	Minimum Full Scale Value (@ +/-10V)	Amp. Gain (tech's use)	Model or Serial No. (tech's use)
Strain Gages				
1. Flywheel Housing, Location #10, leg A	0	1000 ue		
2. Flywheel Housing, Location #10, leg B	1	1000 ue		
3. Flywheel Housing, Location #10, leg C	2	1000 ue		
4. Flywheel Housing, Location #13, leg A	3	1000 ue		
5 Flywheel Housing, Location #13, leg B	4	1000 ue		
6. Flywheel Housing, Location #13, leg C	5	1000 ue		
7. Flywheel Housing, Location #1	6	2000 ue		
Accelerometers				
8. Engine, front-center of oil pan (Vert., V)	7	15 g		
9. Engine, front-center of oil pan (Lat., T)	8	15 g		
10. Flywheel Housing, bottom-center (Vert., V)	9	15 g		
11. Flywheel Housing, bottom-center (Lat., T)	10	15 g		
12. Transmission, bottom-rear-center (Vert., V)	11	15 g		
13. Transmission, bottom-rear-center (Lat., T)	12	15 g		
14. Transfer Case, top-rear-center (Vert., V)	13	20 g		
15. Transfer Case, top-rear-center (Lat., T)	14	20 g		
16. Transfer Case, top-rear-center (Long., L)	15	20 g		
17. Rear Differential, top-front-center (Vert., V)	16	15 g		
18. Rear Differential, top-front-center (Lat., T)	17	15 g		
19. Rear Differential, top-front-center (Long., L)	18	15 g		
20. Front Differential, top-rear-center (Vert., V)	19	15 g		
21. Front Differential, top-rear-center (Lat., T)	20	15 g		
22. Front Differential, top-rear-center (Long., L)	21	15 g		

Miscellaneous:

Engine Speed (rpm)

Dynamometer Front Axle Speed (mph)

Dynamometer Rear Axle Speed (mph)

Dynamometer Front Axle Torque (ft-lbs)
Dynamometer Rear Axle Torque (ft-lbs)

Temperatures:

Transfer Case Oil
Ambient
Tire
Differential Fluid
Rear Driveshaft, Front U-Joint
Front Differential
Rear Differential

Pressures:

Transfer Case Oil

Installation:

A chassis dynamometer will be set up in Building 212's test cell 9. A baseline LMTV will be provided by the PM's office. It's tires will be shaved, and then it will be set up and instrumented on the chassis dynamometer by personnel from TARDEC's mobility area. For added knowledge, a remote video camera will be placed under the vehicle to record and allow visual inspection of the system while the vehicle is in simulated motion.

A - Chassis Dynamometer/Vehicle Resistance Loss:

Objective: To determine the power required to drive the vehicle at each speed so that this can be accounted for when applying torque to the vehicle during testing.

1. Motor the dynamometer and record the power and torque required to propel the vehicle at 5 mph increments up to the limit of the chassis dynamometer.

B – Effect of Runout on Vehicle Vibration

Objective: The objective of this test is to determine what the impact of the balance and runout on a driveshaft is on overall vehicle vibration and u-joint temperature. This will be accomplished by comparing the test results of two rear driveshafts, one which has been re-balanced and straightened, the other which is unaltered and straight off the production line.

1. Install a very well balanced shaft with low runout on the vehicle.
2. Remove all payload from the vehicle bed.
3. Set dynamometer absorption to simulate level paved road.

4. Record accelerometer outputs as a function of vehicle speed (from 20 to 60 mph in 5 mph increments) and other critical data.
5. Add 5000 pounds of load to the vehicle bed.
6. Set dynamometer to simulate level paved road.
7. Record accelerometer outputs as a function of vehicle speed (from 20 to 60 mph in 5 mph increments) and other critical data.
8. Install rear driveshaft, with excessive runout, on vehicle
9. Repeat steps 2 through 7

C - U-joint/Prop Shaft Angle Test:

Objective: The purpose of this test is to determine whether or not the U-joint/prop shaft angle is the cause of increased strain in the drivetrain system. We will do this by testing an LMTV at several different prop shaft angles with production driveshafts. This will allow us to determine a relationship between prop shaft angle and the resultant forces.

High Angle Vehicle Characteristics:

1. Remove all load from the vehicle bed.
2. Prop up vehicle to relieve extra load off of tires
3. Set dynamometer absorption to simulate level paved road.
4. Record accelerometer outputs as a function of vehicle speed (from 20 to max vehicle speed in 5 mph increments) and other critical data.

No Payload Vehicle Characteristics:

1. Remove all load from the vehicle bed.
2. Set dynamometer absorption to simulate level paved road.
3. Record accelerometer outputs as a function of vehicle speed (from 20 to max vehicle speed in 5 mph increments) and other critical data.

Full Payload Vehicle Characteristics:

1. Add 5000 pounds of load to the vehicle bed.
2. Set dynamometer to simulate level paved road.
3. Record accelerometer outputs as a function of vehicle speed (from 20 to max vehicle speed in 5 mph increments) and other critical data.

Low Angle Vehicle Characteristics (if feasible):

1. Add excess payload, or pull down vehicle by clamping, to achieve a minimum angle.
2. Set dynamometer to simulate level paved road.
3. Record accelerometer outputs as a function of vehicle speed (from 20 to max vehicle speed in 5 mph increments) and other critical data.

D - CV-Joint Test:

Objective: The purpose of this portion of testing is to determine if the application of a continuously variable joint (CV-joint) at the same operating angles as a universal joint (U-joint) will reduce the strains and accelerations in the drivetrain to acceptable levels while keeping reasonable CV-joint temperatures.

1. Replace U-Joint with PM provided CV-joint on baseline LMTV.
2. Run the High Angle Vehicle Characteristics test, No Payload Vehicle Characteristics test, Full Payload Vehicle Characteristics test and the Low Angle Vehicle Characteristics test as mentioned above with the CV-Joint installed.

Timeline:

The projected brief-out/report date is August 19, 1998.

APPENDIX B
Miscellaneous Referenced Documents

Author: Rick Agnetti at AMSTAR2POST
Date: 7/29/98 9:29 AM
Priority: Normal
TO: Stephen 'WGM' Aamodt, Julian Kozowyk, Charles Raffa
CC: Roger Olson, Ron Beck at AMSTAR1POST
Subject: Propshaft

Gentlemen:

I just received the report on the rear propshaft of the LMTV vehicle. This vehicle is in Test Cell #9 and has 57 miles on the odometer. Upon inspection the following was noted:

Working Length: 59" / 61 3/4"

Total Slip: 5.5"

** Balance spec's: This contractor typically balances a shaft to a .25 oz.in. Our shaft as is off the vehicle spec'ed 1.7 oz.in. on the slip end and 1.5 oz.in. on the welded yoke end. It was noted that our shaft only had balance weight at one end.

** Roundness: Our shaft was also found to be out of round and hobbling at the slip-joint. The measurements were:

	Left (stub)	Center (Tube)	Right (weld yoke)
In	.007	.011	.008

The contractor stated that this shaft would have never left their facility. The combination of the out of balance and out of round would cause premature wear on the u-joints and could reduce the life of the u-joints up to 70%. This is before adding in the 11 degree angle that it operates at in the vehicle.

The contractor then balanced and straightened the shaft to their acceptable spec's. The results are: .23oz.in. at the slip end and .25oz.in. at the welded yoke end. After straightening the roundness spec's are:

	Left (stub)	Center (Tube)	Right (weld yoke)
Out	.003	.008	.006

The contractor made one final comment about the application and that was if your going to run the shaft at an 11 degree angle then you need to run a perfectly balanced shaft.

I have been directed to have an evaluation done on the front propshaft and will report the results in a day or two.

Martin R. Agnetti
Eng Tech
Team Test Ops

JOINT, CLUTCH & GEAR SERVICE, INC.

SERVING JOBBERS THROUGHOUT MICHIGAN

Bay City Branch
703 MARQUETTE ST.
BAY CITY, MI 48706
Phone (517) 666-2024
Fax (517) 666-9906
In MI (800) 202-1520

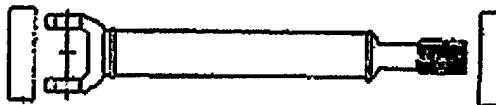
East Branch
22705 HOOVER
WARREN, MI 48089
Phone (313) 755-1515

Main Office
1325 HOWARD STREET
DETROIT, MI 48226
Phone (313) 951-4400
Fax (313) 951-4500
In MI (800) 572-8240

West Branch
6516 N. TELEGRAPH
DEARBORN, MI 48127
Phone (313) 585-7822

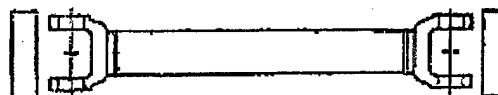
Grand Rapids Branch
4103 STAFFORD, S.W.
WYOMING, MI 49509
Phone (616) 536-7302
Fax (616) 536-9280
In MI (800) 442-8235

BALANCE SPECS



Acceptable Balance Specs are .05 in oz per end

LONG 1.1 .70
SHAFT .60 1.3 Bell Street



Acceptable Balance Specs are .05 in oz per end

Shaft Series _____

Company Name _____

Who Ordered _____

JC&G Job# _____

Your P.O.# _____

Date _____

Distributors of Original Equipment Clutches, Power Take-Offs, Universal Joints & Exhausts

Balance Specs for First Replacement Set of Meritor Driveshafts

JOINT, CLUTCH & GEAR SERVICE, INC.

SERVING JOBBERS THROUGHOUT MICHIGAN

Bay City Branch
703 MARQUETTE ST.
BAY CITY, MI 48708
Phone (517) 686-2924
Fax (517) 686-9906
In MI (800) 292-1520

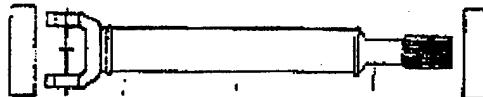
East Branch
22705 HOOVER
WARREN, MI 48089
Phone (510) 756-1515

Main Office
1325 HOWARD STREET
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DEARBORN, MI 48127
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Grand Rapids Branch
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WYOMING, MI 49509
Phone (616) 534-7360
Fax (616) 534-8280
In MI (800) 442-8236

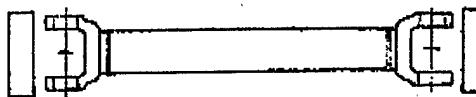
RUNOUT SPECS



Acceptable Runout Specs are left .015 center .015 right .005

Long	.014	.013	.021
Short	.017	.029	.040

Bill Etter



Acceptable Runout Specs are left .015 center .015 right .015

Shaft Series _____

Company Name _____

Who Ordered _____

JC&G Job# _____

Your P.O.# _____

Date _____

Distribution of Original Equipment Clutches, Power Take-Offs, Universal Joints & Extenders

Runout Specs for First Replacement Set of Meritor Driveshafts

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BAY CITY, MI 48708
Phone (517) 626-2924
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In MI (800) 292-1520

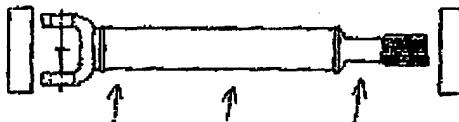
East Branch
22705 HOOVER
WARREN, MI 48090
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Main Office
1325 HOWARD STREET
DETROIT, MI 48226
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In MI (800) 672-6240

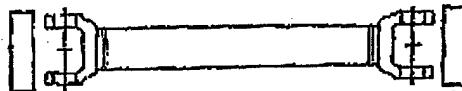
West Branch
8516 N. TELEGRAPH
DEARBORN, MI 48127
Phone (313) 593-7522

Grand Rapids Branch
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WYOMING, MI 49509
Phone (316) 538-7300
Fax (316) 532-6200
In MI (800) 442-4235

RUNOUT SPECS



Acceptable Runout Specs are left .015 center .015 right .005
.075 >.250 .175



Acceptable Runout Specs are left .015 center .015 right .015

Shaft Series

1610

Company Name

U.S. Army

Who Ordered

Rick Agnelli

JC&G Job#

—

Your P.O.#

—

Date

8/13/98

Distributors of Original Equipment Clutches, Power Take-Offs, Universal Joints & Bushings

Data from the Wobbling Shaft

APPENDIX C

Sample Data

TACOM
Run# 387
started 10/2/98
at 13:21

TEST PURPOSE: GO FOR MILES
Model: MODEL # POS#1LMTV

DATE	Wall Time	Elapsed Time	FASPEED mph*	FTORQUE lb-ft	RASPEED mph*	RTORQUE lb-ft	ENG rpm	FLYWHEEL PSI	AIR AMBT °F	FLYWHEEL °F	FLYWHEEL °F	UJOINT °F	TIRE °F	TCASE °F	FCASE °F	TCASE °F	FCASE °F	DIFF R °F	OIL °F	OIL °F	OIL °F
10/02/98	13:21	00:34:25.2	-	1.1	-	0.2	31	-0.04	80.6	65.9	71.5	70.5	2.7	79.6	67.4	69.8	-	-	-	-	
10/02/98	13:57	00:36:28.5	-	13.7	-	18.4	2549	0.03	95.8	70.9	115	107.6	164.9	210.3	135.7	135.6	-	-	-	-	
10/02/98	14:05	00:44:45.7	-	13.8	-	18.8	2572	0.04	104.2	69.9	112	109.6	165.2	241.3	155.8	154.2	-	-	-	-	
10/02/98	14:11	00:50:44.4	-	12.6	-	17.1	2457	0.03	108.6	70.1	112	109.6	164.5	250.8	166.2	163.8	-	-	-	-	
10/02/98	14:11	00:50:45.7	-	12.6	-	17	2458	0.03	108.6	70.1	112	109.6	164.5	250.8	166.3	163.8	-	-	-	-	
10/02/98	14:12	00:51:10.4	-	0.7	-	-0.1	804	-0.04	108.9	70	126	102	268.2	242.2	165.4	163.3	-	-	-	-	
10/02/98	14:25	00:03:58.1	-	1	-	0.1	130	-0.02	109.4	66.3	99.5	90.9	2.1	195.6	133.2	126.8	-	-	-	-	
10/02/98	14:50	00:12:11.6	-	12.6	-	17.4	2503	0.03	108.3	67.7	113	105.5	165	222.6	150.7	149.9	-	-	-	-	
10/02/98	14:51	00:13:03.2	-	12.4	-	17.4	2505	0.03	108.6	67.8	115	107.3	165	226.2	152.6	151.8	-	-	-	-	
10/02/98	14:53	00:13:15.6	-	12.3	-	17.6	2504	0.03	110	68.7	115	108.7	164.9	233	156.5	155.9	-	-	-	-	
10/02/98	15:00	00:13:17.1	-	12.4	-	17.6	2503	0.03	113.9	68.8	120	110.5	164.4	248.2	168.5	168	-	-	-	-	
10/02/98	15:21	00:20:00.9	-	6.4	-	5.3	2602	0.03	119.6	69.9	117	114.9	163.8	263.3	188.6	188.6	-	-	-	-	
10/02/98	15:24	00:20:33.7	-	6.4	-	5.2	2599	0.03	119.7	69.5	119	115.2	163.8	264.6	190.5	190.9	-	-	-	-	
10/02/98	15:28	00:20:33.6	-	6.4	-	5	2590	0.03	120.2	69.9	118	115.5	163.3	265.4	192.6	193.4	-	-	-	-	
10/02/98	15:50	00:22:35.1	-	6.3	-	4.8	2583	0.03	121.5	70.6	121	115.5	163.2	265.2	199.7	203.3	-	-	-	-	
10/02/98	15:50	00:22:36.1	-	6.4	-	4.8	2583	0.03	121.7	70.6	121	115.8	163.3	265.1	199.7	203.3	-	-	-	-	
10/02/98	15:50	00:22:37.2	-	6.3	-	4.8	2583	0.03	121.6	70.6	121	115.8	163.3	265.1	199.7	203.4	-	-	-	-	
10/02/98	15:51	00:23:00.9	-	1.5	-	0.5	798	-0.04	121.7	70.7	125	114.3	166.2	263	199.2	202.4	-	-	-	-	
10/02/98	16:21	00:30:00.9	-	0.8	-	0	31	-0.04	113.6	67.7	90.6	83.6	2.1	182.9	125.4	131	-	-	-	-	
10/02/98	16:50	00:32:25.0	-	5.9	-	4.5	2526	0.03	113.1	68.9	118	107.9	164.1	240.6	165.9	169.4	-	-	-	-	
10/02/98	16:51	00:33:00.9	-	5.9	-	4.5	2525	0.03	113.5	68.6	119	108.5	164.1	241.9	166.7	170.2	-	-	-	-	
10/02/98	16:55	00:34:43.9	-	5.9	-	4.5	2526	0.04	115.3	69.3	117	109.6	163.7	249.4	173.2	175.7	-	-	-	-	
10/02/98	16:57	00:35:47.9	-	5.9	-	4.5	2526	0.03	115.6	69.5	118	110.5	163.7	250.5	174.4	176.8	-	-	-	-	

10/02/98	17:18	003:57:37.2	-	5.9	-	4.6	2525	0.03	119	70.1	120	112.
10/02/98	17:18	003:57:38.8	-	5.9	-	4.7	2525	0.03	119	70	120	112.
10/02/98	17:18	003:57:40.9	-	6	-	4.7	2526	0.03	119	70	120	112.
10/02/98	17:19	003:57:47.0	-	5.9	-	4.7	2525	0.02	119	69.9	119	112.
10/02/98	17:19	003:57:52.9	-	5.9	-	4.6	2524	0.03	119	69.9	119	112.
10/02/98	17:19	003:57:56.7	-	5.9	-	4.7	2525	0.03	119	69.9	120	112.
10/02/98	17:19	003:58:01.7	-	5.9	-	4.6	2527	0.03	119	69.9	120	112.
10/02/98	17:19	003:58:08.7	-	5.9	-	4.5	2525	0.03	119	69.8	120	112.
10/02/98	17:19	003:58:17.3	-	5.8	-	4.5	2526	0.03	119.1	69.8	120	112.
10/02/98	17:19	003:58:27.5	-	5.9	-	4.7	2526	0.03	119	69.7	120	112.
10/02/98	17:19	003:58:37.4	-	5.9	-	4.5	2526	0.03	119	69.5	120	112.
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10/02/98	17:20	003:58:48.4	-	5.8	-	4.7	2526	0.03	119.2	69.7	120	112.
10/02/98	17:20	003:58:49.3	-	5.8	-	4.6	2525	0.03	119	69.4	120	112.
10/02/98	17:46	004:25:32.5	-	6	-	4.7	2588	0.03	121.4	69.7	120	114.
10/02/98	17:46	004:25:34.1	-	5.9	-	4.7	2590	0.03	121.4	69.8	120	114.
10/02/98	17:49	004:28:46.0	-	6	-	4.8	2600	0.03	121.8	70.4	121	114.
10/02/98	17:50	004:28:58.9	-	5.9	-	4.7	2589	0.03	121.8	70.2	121	114.
10/02/98	18:07	004:46:42.9	-	6.4	-	4.9	2589	0.03	121.6	69.3	121	11
10/02/98	18:07	004:46:44.0	-	6.3	-	5	2590	0.04	121.6	69.4	121	11
10/02/98	18:08	004:47:27.4	-	6.1	-	4.9	2590	0.03	121.6	69.7	122	11

* - for this test speed was measured by correlating engine speed and transmission gear ratio to achieve approximately 57 mph